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# FLAME TEMPERATURES IN AN INTERNAL COMBUSTION ENGINE MEASURED BY SPECTRAL LINE REVERSAL

BY

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ENGINEERING EXPERIMENT STATION

BULLETIN No. 262 OCTOBER, 1933

# FLAME TEMPERATURES IN AN INTERNAL COMBUSTION ENGINE MEASURED BY SPECTRAL LINE REVERSAL

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# FLAME TEMPERATURES IN AN INTERNAL COMBUSTION ENGINE MEASURED BY SPECTRAL LINE REVERSAL

## I. Introduction

1. *Object of Investigation.*—A fuel burning in the cylinder of an internal combustion engine forms a gaseous mixture whose properties, although not uniform, may be assumed to vary continuously throughout the combustion space. Such a mixture constitutes a homogeneous system and its thermodynamic state at any instant can be completely determined only when suitable values can be assigned to the variables $p$, $V$, $T$, $c_1$, $c_2$, . . . . . $c_n$ (pressure, volume, temperature, and gas concentrations) either by direct measurement or by calculation. Methods for measuring instantaneous values of $p$ and $V$ are well developed, but independent determinations of either the instantaneous temperatures or concentrations are considerably more difficult. The concentrations of some of the gaseous components have been determined, using a sampling valve located in different parts of an engine cylinder*; but the experimental methods employed in making nearly all previous measurements of the gas temperatures in an engine† are open to criticism and are entirely inadequate for engine speeds above 500 r.p.m. Since knowledge of the temperature is of major importance, it seemed desirable to attempt direct temperature measurement by an optical method capable of following the rapid fluctuations which occur in the engine, but not affecting the combustion process itself. The method of line reversal, which has been used extensively in the determination of the temperature of stationary flames, seemed to meet these requirements‡ and has, therefore, been used to measure the instantaneous flame temperatures during combustion in a spark ignition engine operating under moderate compression and air-fuel ratios varying from 9:1 to 20:1. This investigation is a continuation of the study of combustion and explosion phenomena carried on by the Engineering Experiment Station over a period of years.§

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*Withrow, Lovell, and Boyd; Ind. and Eng. Chem. Vol. 22, p. 945 (1930).

†Hopkinson: Phil. Mag., Vol. 13, S-VI, p. 84 (1907).
Callendar and Dalby: Proc. Royal Soc., Vol. 80, p. 57 (1908)
Coker and Scoble: Proc. Inst. of Civil Eng., Vol. 196, p. 1 (1913)

‡The results of another optical method used by one of the authors in a study of the same problem are given in Ind. and Eng. Chem., Vol. 24, p. 867 (1932).

§"A Study of Explosions of Gaseous Mixtures," Univ. of Ill. Eng. Exp. Sta. Bul. No. 133, 1922.
"An Investigation of the Mechanism of Explosive Reactions," Univ. of Ill. Eng. Exp. Sta. Bul. No. 157, 1926.
"A Thermodynamic Analysis of Internal Combustion Engine Cycles," Univ. of Ill. Eng. Exp. Sta. Bul. No. 160, 1927.

2. *Acknowledgments.*—The authors wish to express their appreciation for the excellent facilities generously made available to them by DEAN M. S. KETCHUM, Director of the Engineering Experiment Station, and PROF. A. C. WILLARD, Head of the Department of Mechanical Engineering, where the work was performed. Personal acknowledgment is due PROF. A. P. KRATZ for his ready assistance and constant interest.

## II. COMBUSTION IN AN ENGINE

3. *Thermodynamic Variables p, V, T, $c_1$, $c_2$ . . . . . $c_n$.*—Strictly speaking, measurement of any thermodynamic property of a system may be made, and a definite meaning attached to the result, only when the system investigated is in complete equilibrium. Obviously gases burning turbulently in an engine are not in complete equilibrium and the only quantity that can be determined with physical certainty is the total volume occupied by the gases at any instant. If any meaning is to be attached to the pressure, temperature, and the several concentrations it is that of an effective average throughout the total volume as recorded by some measuring device. This recorded value will be influenced always by the properties of the measuring instrument. Measurements of pressure, using any one of several modern high speed indicators, give such effective values, since pressure equilibrium is established throughout the region at velocities comparable to that of sound. This velocity is far above that of the piston and the assumption is quite justified that such determinations of pressure give values which may be interpreted correctly as characteristic of the effective pressure equilibrium existing at any given instant.

Measurements of the concentrations, $c_1$, $c_2$ . . . . . $c_n$, of the $n$ constituents in the combustible at different phases of the cycle are open to more definite questioning than the pressure measurements, for several reasons. In an actual engine the mixture cannot be uniform when ignition occurs; and the normal flame velocities, though much higher than the piston velocity, are less than that of sound.* Hence the concentrations determined by any sampling valve method, if averaged, will give effective values which will have a wider range of uncertainty than that of the corresponding pressure measurements. Also, the samples themselves will be removed from a rapidly burning mixture at high temperatures and of necessity analyzed at some later time

*"Flame Movement and Pressure Development in an Engine Cylinder," N.A.C.A. Report No. 399, 1931.

at lower temperatures with no means available for stopping the reaction on the instant the sample is removed. Certainly concentration measurements can give only a very approximate picture of conditions in the engine during the time of burning.

Whether sufficiently uniform conditions exist in the cylinder of an engine to justify assigning any thermodynamic significance to measured temperatures is difficult to decide. Conduction and natural convection processes are far too slow compared to the piston velocity to do much toward smoothing out temperature gradients in the cylinder. Diffusion must also be too slow; for although the velocities of the particles are extremely high, the densities are likewise, so that the mean free path will always be small. However, transfer of heat energy by radiation processes will be very rapid, since the velocity of travel is that of light, and the time any particle can retain its absorbed energy of the order of $10^{-8}$ seconds.* Therefore, the effective velocity for radiation transfer will be very high. Furthermore, the CO, $CO_2$, and $H_2O$ molecules which are present in the products of combustion are known to be effective absorbers and radiators in the infra red region where the radiant energy has its maximum. Hence it seems reasonable to expect radiation processes to play a major role in establishing thermal equilibrium throughout the cylinder at any instant. Certainly such processes, perhaps aided somewhat by turbulence, are the only means whereby thermal equilibrium can be approached in times relatively short compared to the time it takes the piston to move through a very small displacement.

If anything approaching instantaneous thermal equilibrium exists, it is important to have a method of measuring temperatures, the response of which will be instantaneous, if the observed results are to be worthy of study. Temperature readings obtained by such a method will indicate whether thermal equilibrium is maintained in the cylinder and, if definite, that radiation processes are largely responsible for that equilibrium. Most methods of measuring temperature are very sluggish, however, and quite incapable of following the fluctuations in the cycle. Besides this, the measuring instrument will distort the conditions in its immediate vicinity by its own heat capacity and so give incorrect results. Callendar and Dalby,† using a light platinum resistance thermometer, were obliged to shield it from the temperature of the flame during the initial stages of combustion to prevent fusion, and were able to take temperature readings near the end of expansion only. They extrapolated back to peak temperatures,

*Kenty: Phys. Rev., Vol. 42, p. 823 (1932).
†Loc. cit., p. 5.

assuming the gases in the cylinder to behave as a gas thermometer. But such extrapolation is open to criticism, since the concentrations of these gases are changing and uncertain.

To obtain the instantaneous values of the effective temperature the thermometric substance must be distributed throughout the entire region where the temperature is desired, and must have a heat capacity negligible as compared with that of the working substance. These conditions can be satisfied by introducing in very small quantities, compared with the total amount of gas present, a thermometric substance which is easily vaporized. The temperature of this substance can then be found by comparing its brightness at any given wave-length with that of a continuous radiator whose brightness temperature at that wave-length is known. The amount of vapor present must be sufficient to radiate and absorb light according to Kirchhoff's Law, and this condition can be attained with extremely small quantities of vapor if comparison is made at a wave-length corresponding to some resonance radiation of the vapor. Under these circumstances, investigation has shown* that the thermometric substance has a negligible effect on the flame temperature. So also in the engine no effect on combustion was observed and the temperature thus determined may be interpreted as the effective temperature at the instant of comparison.

When such effective temperatures are known, corresponding concentrations can be computed, assuming chemical equilibrium. These concentrations will likewise be effective values and, while their uncertainty will be at least as great as that of the temperature measurements, they should indicate in a general way the progress of the chemical reactions.

4. *Temperature Measurement by Line Reversal.*—The method of temperature measurement just outlined is known as the line-reversal method, and has been used extensively in measuring the temperatures of a wide variety of flames.† The principles involved can be readily understood by considering the application of the method in the determination of stationary flame temperatures. In Fig. 1, $F$ represents a bunsen flame into which a metallic vapor such as sodium is introduced. The vapor must be that of an element whose resonance radiation is in a wave-length region which can be easily observed. Radiation from the tungsten ribbon filament lamp at $L$ passes

*Jones, Lewis, Friauf, and Perrott: Journ. Amer. Chem. Soc., Vol. 53, p. 869 (1931).
†Loc. cit., p. 8.
Henning and Tingwalt: Z. Physik, Vol. 48. p. 805 (1928).
Loomis and Perrott: Ind. and Eng. Chem., Vol. 20, p. 1004 (1928).
Griffiths and Awberry: Proc. Roy. Soc., Vol. 123, p. 401 (1929)

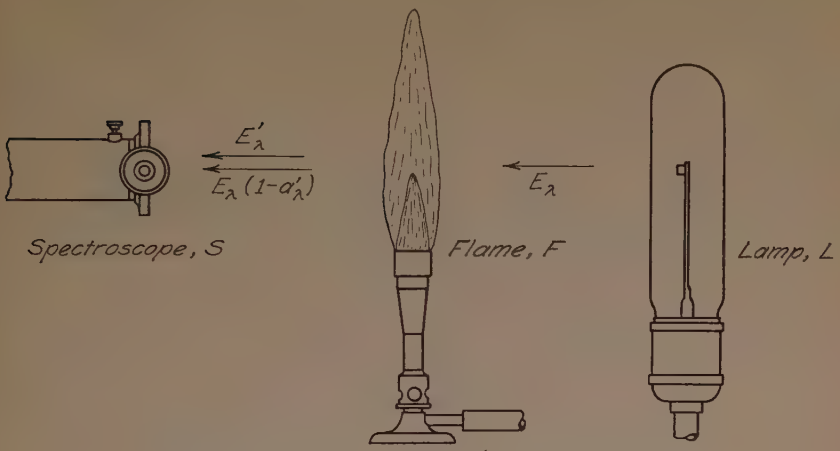


FIG. 1. DIAGRAM OF LINE REVERSAL APPARATUS

through the flame to the slit of the spectroscope at $S$. The flame is practically transparent to radiation in the visible region except at $\lambda$, the wave-length of the resonance radiation of the metallic vapor in the flame. At this wave-length, due to the presence of the vapor, strong absorption as well as emission occur. Let $E_\lambda$ and $E'_\lambda$ be the monochromatic emissive power at the wave-length $\lambda$ for the lamp filament and the flame respectively, and let $a'_\lambda$ be the absorptivity of the flame at the same wave-length. The energy which reaches the slit of the spectroscope at this wave-length will be $E'_\lambda + (1 - a'_\lambda)E_\lambda$, assuming that reflection at the flame surface is negligible. If

$$E'_\lambda > a'_\lambda E_\lambda$$

a bright line will be observed, crossing the continuous spectrum of the filament radiation at $\lambda$. If

$$E'_\lambda < a'_\lambda E_\lambda$$

a dark or reversed line will appear at this wave-length. For the particular case when

$$E'_\lambda = a'_\lambda E_\lambda \qquad (1)$$

neither the bright nor the reversed line will be seen, and the flame will be emitting as much energy of wave-length $\lambda$ as it is absorbing.

If the flame satisfies Kirchhoff's Law for radiation of wave-length $\lambda$,* then

$$\frac{E'_\lambda}{a'_\lambda} = \mathbf{E}_\lambda \tag{2}$$

where $\mathbf{E}_\lambda$ is the monochromatic emissive power of a perfect radiator at the wave-length $\lambda$, the true temperature of this radiator being the same as that of the vapor in the flame. Therefore,

$$E_\lambda = \mathbf{E}_\lambda \tag{3}$$

From Wien's Equation

$$E_\lambda = \frac{C_1}{\lambda^5} \cdot \exp.\left(-\frac{C_2}{\lambda S_\lambda}\right) \tag{4}$$

where $S_\lambda$ is the brightness temperature of the lamp filament at the wave-length $\lambda$. Also

$$\mathbf{E}_\lambda = \frac{C_1}{\lambda^5} \cdot \exp.\left(-\frac{C_2}{\lambda T'}\right) \tag{5}$$

where $T'$ is the true temperature of the vapor in the flame. Hence

$$T' = S_\lambda \tag{6}$$

If it is assumed that the vapor is in thermal equilibrium with the gases in the flame, and that its radiation is the result of purely thermal excitation, then the brightness temperature $S_\lambda$ of the lamp filament must be the same as the true flame temperature. The agreement between the values found for the temperature of stationary flames by the line-reversal method and by other independent methods† is satisfactory proof of the validity of these assumptions.

The brightness temperature of the lamp filament can be measured with an optical pyrometer, so that the true temperature of the flame can be readily determined. However, when the effective wave-length of the pyrometer screen is not the same as that of the resonance radiation of the vapor in the flame, it is necessary to make a slight correction for the variation in emissivity of the lamp filament between these two wave-lengths. If $\epsilon_\lambda$ is the monochromatic emissivity of the filament at wave-length $\lambda$ then

*Kohn: Ann. der Physik, Vol. 44, S. 14, p. 749, (1914) has shown that this is true for stationary flames colored with alkali-metal vapors.

†Jones, Lewis, Friauf and Perrott: Jour. Amer. Chem. Soc., Vol. 53, p. 869 (1931).

$$E_\lambda = \epsilon_\lambda \cdot \frac{C_1}{\lambda^5} \cdot \exp.\left(-\frac{C_2}{\lambda T}\right), \tag{7}$$

when $T$ is the true filament temperature. Combining this equation with (4) gives

$$\frac{1}{T} - \frac{1}{S_\lambda} = \frac{\lambda}{C_2} \ln. \epsilon_\lambda \tag{8}$$

Evaluating this equation for the two wave-lengths, and eliminating $T$, gives, finally,

$$\frac{1}{S_{\lambda_2}} = \frac{\lambda_1}{C_2} \ln. \epsilon_{\lambda_1} - \frac{\lambda_2}{C_2} \ln. \epsilon_{\lambda_2} + \frac{1}{S_{\lambda_1}} \tag{9}$$

where $\lambda_1$ is the effective wave-length of the pyrometer screen and $\lambda_2$ is the wave-length of the radiation from the vapor. The pyrometer measurement gives the value of $S_{\lambda_1}$, the constant $C_2$ is the same for all temperatures, the emissivity for a tungsten filament is known over a wide range of wave-lengths and temperatures, and hence the value of $S_{\lambda_2}$, which is equal to the true flame temperature, can be calculated from Equation (9).

In measuring the flame temperature in an internal combustion engine by the line-reversal method it is only necessary to pass the lamp radiation through the flame in the engine by means of suitable windows on opposite sides of the cylinder, and to introduce sufficient sodium into the combustible to make it possible to observe both the bright and the reversed line. A stroboscopic shutter, driven from the engine crankshaft, limits the observation to a short interval in successive cycles and thus gives an approximation to the instantaneous values of the effective flame temperature.

## III. Description of Apparatus and Methods of Calibration

5. *Engine.*—The engine of a farm lighting plant was used for the experimental work. In order to obtain satisfactory control of operating conditions, the original air-cooled cylinder and head were replaced with special water-cooled units. The design of these new parts was such as to insure adequate and uniform cooling to all parts of the combustion space and prevent the formation of hot spots. The cylindrical combustion chamber was 3¼ inches in diameter, this being the bore of the engine cylinder. The jacket space stopped 1⅜ inches

from the top of the cylinder, so that a solid ring was left in which four equally spaced 7/8-inch holes were drilled. In two of these holes, on opposite sides of the combustion chamber, the cylinder windows were placed. These windows were of quartz and were held in shells which were screwed into the cylinder so that the inner surfaces of the windows were flush with the surface of the combustion chamber. The valves were located in the flat cylinder head, thus providing a combustion space which could be completely machined and which contained no cavities. The height of the chamber could be adjusted from 1.000 inch to 1.667 inches by means of rings of varying thickness placed between the cylinder and head, thus changing the compression ratio from 6:1 to 4:1.

A one-inch Stromberg plain tube carburetor was used, but with several modifications. Since the engine was operated within a narrow speed range and with wide open throttle, only the main jets were used and the idling jet was completely closed. The needle valve for the main jets was replaced with one having a more gradual taper and a large graduated head to give finer adjustment. With the air-metering system employed, the pressure at the carburetor inlet was approximately 2 in. of water below atmospheric, so that it was necessary to seal the float chamber and connect it and the air bleed to the air inlet pipe. The capacity of a one-inch carburetor being considerably in excess of the demands of an engine having a displacement of 41 cubic inches, the large venturi tube had to be replaced by one with a smaller bore. Since the mixture could be enriched with the main needle valve for starting, the choke valve was removed.

The spark plug was located in the cylinder head just above one of the cylinder windows, and, due to space limitations, a 14-millimeter plug was used. The breaker cam was fastened directly to the camshaft, while the case carrying the breaker arm could be securely clamped to prevent any variation of spark timing when the engine was running. The current source was a potential divider across the 110-volt d.c. power line which supplied a 12-volt ignition coil.

The engine was loaded by means of the direct-connected 32-volt generator with which the power unit was equipped. By connecting the generator to the 110-volt d.c. power line with suitable resistance, it also served as a motor for starting and for motoring tests. The generator field, which was originally self-excited, was also connected to the d.c. supply for starting and for independent voltage control when loading the engine.

6. *Engine Control and Metering Apparatus.*—Of the numerous factors affecting combustion in an engine, the following were selected as being the most important and, therefore, requiring the closest control:

(a) Air–Fuel Ratio
(b) Compression Pressure
(c) Inlet Temperature
(d) Jacket Temperature
(e) Spark Timing
(f) Speed
(g) Fuel Characteristics

For this particular series of tests the air–fuel ratio was the variable while the remaining factors were held as nearly constant as possible. A few tests with constant air–fuel ratio and varying spark timing were also made.

**(a) Air–Fuel Ratio**

The air consumption of the engine depends only on engine speed and throttle position, and hence the rate was constant, since the load was adjusted to give a constant speed, and the throttle was always wide open. The fuel consumption, therefore, determined the air–fuel ratio and was controlled by the setting of the carburetor needle valve. The air and fuel metering apparatus are shown in Fig. 2. The air was measured with an Emco No. 1 Dry Test Meter. Between it and the carburetor inlet was an expansion chamber with a volume 52 times the engine displacement. This served to reduce the pressure fluctuations at the meter to less than 0.1 in. of water. The drop in pressure through the meter was measured by means of a manometer connected to a piezometer ring at the outlet, and was found to be about 2 in. of water at operating rates. A thermometer indicated the temperature of the air leaving the meter. The temperature being known, and the pressure in the bellows assumed to be less than atmospheric by half the total drop through the meter, the density of the air could be determined.

The meter was calibrated under actual operating conditions of pulsating flow by allowing the engine to pump a known weight of air through it from the air-weighing tank.* Atmospheric pressure was maintained at the meter inlet by adjusting the throttling valves of the air-control manifold and the engine was operated at constant speed. From Fig. 3, in which the results of the calibration are shown graphically, it is evident that the correction is independent of the

*See Engineering Experiment Station Bulletin No. 207 for a description of the air weighing equipment.

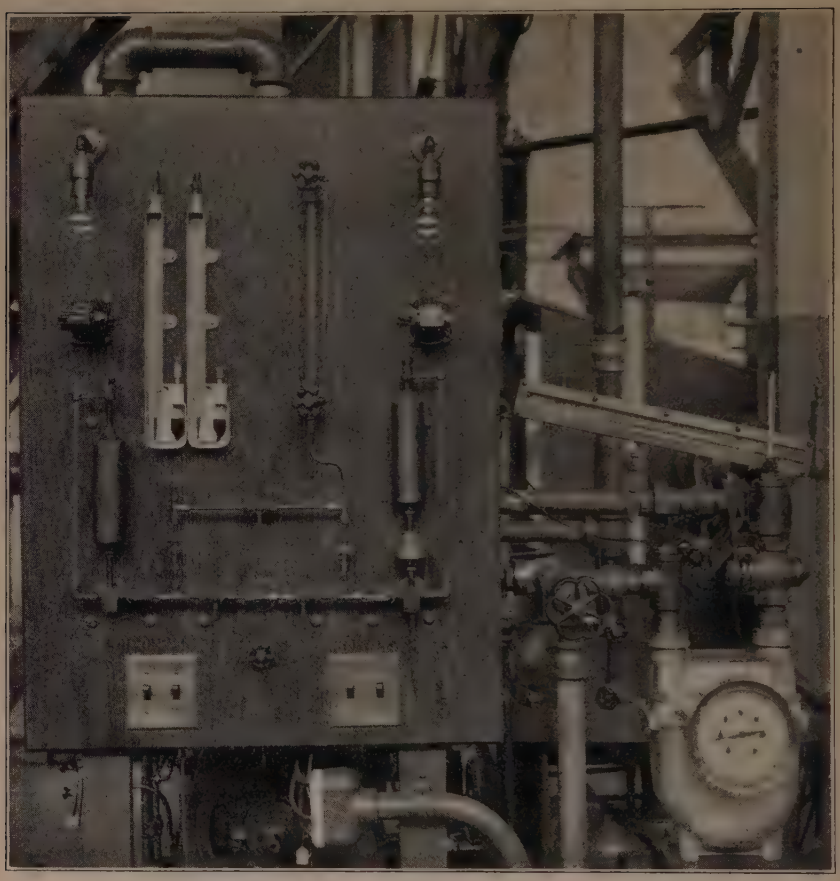

FIG. 2. FUEL AND AIR METERING APPARATUS

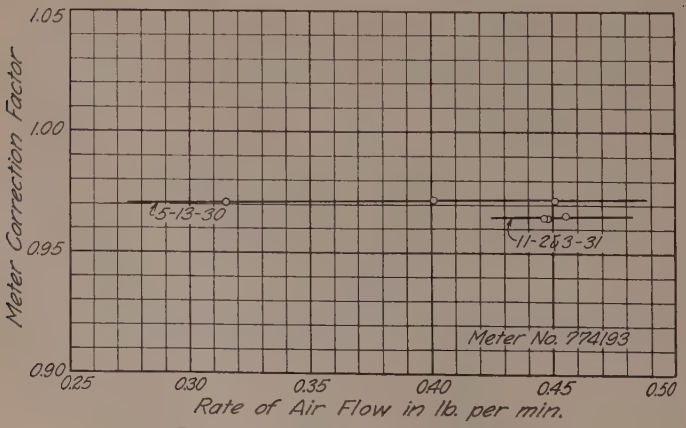


FIG. 3. AIR METER CALIBRATION CURVE

rate of flow. The meter was found to read about 3 per cent high and, since such meters can generally be relied upon for an accuracy of at least 1 per cent, this greater error was probably caused by the pulsating flow. Calibrations were made at the beginning and end of the tests with an interval of more than a year between, and the change in the meter correction factor was from 0.971 to 0.964, or 0.7 per cent.

The fuel was measured volumetrically by means of the small tanks shown in Fig. 2. Using a fuel of known density, the volume of these metering tanks was found, and the weight of a measured amount of any fuel could then be found, once its density was known. In calibrating the tanks a chemical balance was used to weigh their contents and a Westphal balance was used for all density determinations.

### (b) Compression Pressure

The compression pressure depends upon the compression ratio and the density of the incoming charge. The compression ratio was unchanged throughout the tests, being determined by the volume of the space between the cylinder and piston head. The charge density was controlled by the inlet temperature, since the volumetric efficiency was constant, the engine being operated at constant speed with wide open throttle, and the change in atmospheric pressure had a negligible effect.

### (c) Inlet Temperature

The inlet temperature was controlled by a jacket around the inlet manifold between the carburetor and cylinder through which hot water from the cylinder jacket could be circulated in varying amounts. No attempt was made to determine the exact temperature of the combustible passing through the manifold, but a thermometer at the end of the manifold jacket served as an indicator for duplicating inlet temperature conditions.

### (d) Jacket Temperature

The cooling water entered the cylinder jacket at the bottom and passed through the solid ring at the top of the cylinder by means of eight equally spaced ½-inch holes into the cylinder head. Part of the hot water leaving the outlet in the cylinder head returned to the inlet in the cylinder where it was recirculated by means of an injector through which the cold water entered the jacket. The advantages of this arrangement are more rapid circulation, small temperature difference between the water inlet and outlet, and small heat capacity in the cooling system. A thermometer in the outlet in the cylinder head

indicated the jacket temperature, which was controlled by means of a valve in the supply line.

**(e) Spark Timing**

The spark timing was controlled in the usual manner by shifting the breaker housing carrying the breaker points. For the earlier tests the spark indicator incorporated in the timer for the balanced diaphragm pressure indicator was used, but later a spark indicator of similar design was mounted on the crankshaft of the engine to provide for more accurate spark setting. For all tests with constant spark timing 36 deg. advance was used, this being the timing which gave the maximum power with the theoretical air–fuel ratio.

**(f) Speed**

The throttle position being fixed, the speed was controlled entirely by changing the load on the engine. This was done by means of a rheostat in the armature circuit of the generator. An Elgin tachometer gave speed indication while operating conditions were being adjusted, but during a test the speed was determined from the total revolutions and the elapsed time occurring while the fuel metering tank was being emptied. A breaker operated by a cam on the engine camshaft, interrupting the circuit of a magnetic counter, constituted the revolution counter, while a manually-operated stop watch measured the time.

**(g) Fuel Characteristics**

The desirable fuel characteristics were: fairly high volatility to insure reasonably homogeneous combustible mixtures at both low and high air–fuel ratios, and a minimum amount of gum to prevent excess carbon deposits on the cylinder windows. A commercial grade of aviation gasoline met these requirements satisfactorily. The fuel was stored in a barrel which was kept filled with nitrogen to minimize any changes in its unsaturated components. The curves in Fig. 4 are the results of A.S.T.M. distillations made at intervals during the tests and they indicate satisfactory fuel stability. The ultimate analysis of the fuel showed the carbon and hydrogen percentages to be in reasonably close agreement with the corresponding percentages in octane, $C_8H_{18}$, which was the formula assumed for all combustion calculations.

7. *Temperature-Measuring Apparatus.*—In general, the necessary apparatus for making line-reversal temperature measurements on a flame consists of a source of continuous radiation, a pyrometer for determining its temperature, a spectroscope, and an aspirator for intro-

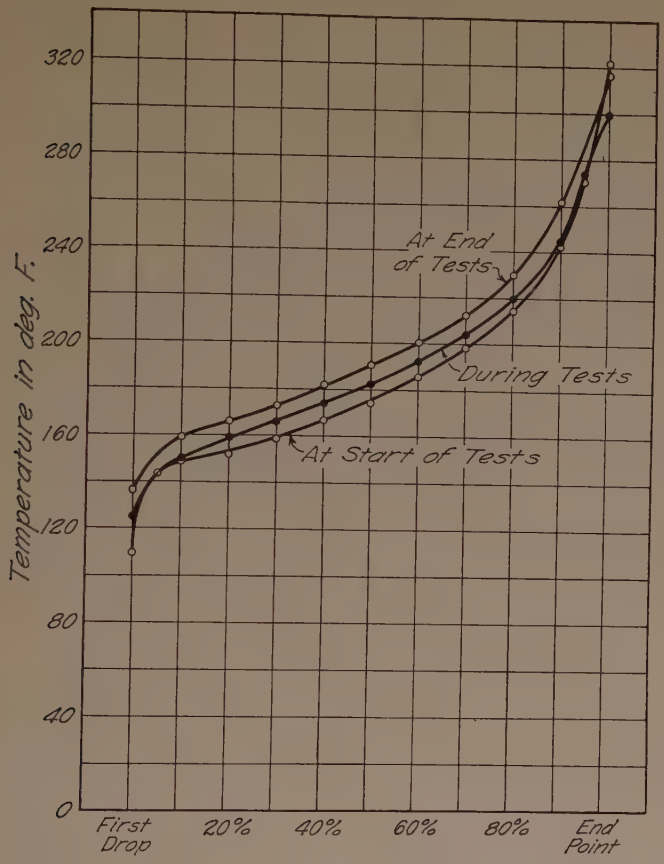


FIG. 4. FUEL A.S.T.M. DISTILLATION CURVES

ducing the metallic vapor into the flame. In the first few engine tests a tungsten ribbon-filament lamp was used as a source; but the high intensity required, together with the vibration of the engine, caused the lamp to fail after only a few hours' use. Another lamp with a V-shaped ribbon filament* supported by a Julius suspension operated satisfactorily during the remainder of the tests. In order to measure the maximum temperatures inside the cylinder, a brightness temperature of the lamp filament at 0.589 $\mu$ of about 4800 deg. F. abs. was necessary. Tungsten has an emissivity at this wave-length and temperature of 0.433 so that the corresponding true temperature of a

*This lamp was one of several made by Mr. Colbey of the Physics Department Shops.

ribbon filament would be 5285 deg. F. abs.* By using a filament with a V-shaped fold the true temperature for the same brightness temperature is much lower since the emissivity of such a cavity is a characteristic of the reflectivity of the filament and the angle of the V. Mendenhall† has shown that the emissivity of a wedge-shaped cavity is

$$\epsilon_\lambda = (1 - r_\lambda^n) \tag{10}$$

where $r_\lambda$ is the reflectivity of the surface for the wave-length $\lambda$, and

$$n = \frac{180}{\alpha} \tag{11}$$

$\alpha$ being the angle of the wedge. For tungsten

$$r_{0.589} = 0.50$$

and with an angle of 60 deg.

$$\epsilon_{0.589} = 0.875$$

This gives a true temperature of 4871 deg. F. abs. for a brightness temperature of 4800 deg. F. abs., or a reduction of 400 deg. F. as compared with the straight ribbon filament. Also, since the reflectivity $r_\lambda$ does not change appreciably in this wave-length region, the brightness temperature at the wave-length of the sodium line, $\lambda = 0.589\ \mu$ is the same as that at the wave-length of the pyrometer screen, or for $\lambda = 0.665\ \mu$, and the correction outlined in discussing the method need not be applied when a V-filament source is used. Filaments with the angle $\alpha$ less than 60 deg. were tried, and, while their emissivities were even higher, they failed before reaching sufficiently high temperatures due to an arc forming across the opening of the V.

The brightness of the lamp filament was controlled by varying the current through it by means of a rheostat in the lamp circuit. Since it was inconvenient to measure the brightness temperature of the filament when the spectroscope was being used, the lamp current was determined whenever a balance between the flame and lamp radiation had been obtained. The corresponding brightness temperatures were then found by calibrating the lamp at the completion of each test. The lamp was calibrated by measuring the brightness temperature for

*See Forsythe and Worthing: Astrophys. Journ., Vol. 61, p. 146 (1925) for radiation characteristics of tungsten filaments.

†Astrophys. Journ., Vol. 33, p. 91 (1911).

a number of different currents and plotting these results to determine a calibration curve. A Leeds and Northrup optical pyrometer was used in measuring the brightness temperature, and the current was measured by means of a potentiometer across a standard 75-ampere ammeter shunt in the lamp circuit.

The optical pyrometer was checked against a standard ribbon-filament lamp, and was found to be in satisfactory agreement. This standard lamp had been calibrated by Dr. W. E. Forsythe at the Nela Park Laboratory of the General Electric Company, and was used as a primary standard for all temperature measurements. In measuring the temperature of the V-filament lamp, however, the pyrometer objective was replaced by the lens which focused light on the spectroscope slit, and hence it was necessary to compare the pyrometer with the standard lamp using this lens. When the same lens was used with both the pyrometer and the spectroscope, the V-filament lamp could be calibrated without changing the optical system, it being only necessary to interchange the two instruments. This simplification saved considerable time since the lamp was calibrated after every test. The pyrometer with this lens was found to read about 10 deg. low at 3180 deg. F. abs. and, since this represents an error of only 0.3 per cent, no corrections were made.

A small Krüss spectrometer with a glass prism was used, and being of very rigid construction, it was well adapted for this particular work. The engine subjected the entire optical system to considerable vibration, but, by clamping the spectrometer to a table secured to the engine foundation, satisfactory optical alignment was maintained. It was of vital importance that this alignment remain unchanged during a test, because only the V of the filament was free from temperature gradients, and this area of the image, which was relatively small, had to fall upon the slit of the spectroscope. The bottom of the V, which appeared as a dark line across the image, acted as a convenient reference line when setting up either the spectroscope or the pyrometer.

The sodium in the form of a NaOH solution was introduced into the engine manifold at the elbow just above the carburetor. A glass aspirator formed a fine spray of the solution which could be picked up and carried into the combustion chamber by the air stream in the manifold. The air for aspirating the solution was drawn from the surge tank in the carburetor air supply line and was, therefore, metered along with the main air supply for the engine. The aspirator outlet was connected to the engine manifold with rubber tubing

so that a pinch cock could be used to control the amount of solution entering the engine cylinder.

A line drawing of the optical system is shown in Fig. 5. The lamp was placed at the principal focus of the first lens and the slit of the spectroscope at that of the second lens. Thus, a parallel beam of light passed through the engine cylinder. This represented a slight modification of the optical system used in measuring the temperature of a stationary flame by line-reversal. In the latter case, the lamp and flame are placed at conjugate foci of the first lens while the flame and slit are at conjugate foci of the second lens. The foci being those which give equal object and image size, a full-sized image of the lamp filament is produced in the middle of the flame by the first lens, while the second lens refocuses this image on the slit. The light passes through the flame in a narrow conical beam and the temperature found is characteristic of a very small region of the flame. In the engine, however, the desired temperature was the effective temperature which would be characteristic of the entire region at any instant, and accordingly, a parallel beam of light of maximum cross section, which would traverse a larger portion of the region, was used instead of a small conical beam.

The temperature of the gases inside an engine cylinder is fluctuating rapidly, and in order to obtain approximately instantaneous values of the temperature at different points in the operating cycle a stroboscopic shutter was used to limit the interval during which radiation from the lamp and flame reached the slit of the spectroscope. The shutter operated at half the crank-shaft speed and had an angular opening of 1¼ deg. However, it was located just outside the cylinder window where it moved across the parallel beam of light which passed through the cylinder, and thus it allowed radiation to reach the spectroscope during considerably more than 2½ deg. of crank-shaft rotation. Moving the shutter by means of its phase-changing gear while the engine was stationary, it was found that light reached the eye-piece of the spectroscope for an interval corresponding to about 18 deg. of crank rotation. In several tests, made to determine the effect of the length of shutter opening on the observed temperatures, the optical system was changed so that the shutter was in the focal plane of the first lens. With this arrangement a shutter opening of one degree allowed light to reach the eye-piece during three degrees of crank rotation.

An aluminum disc 6 inches in diameter, with an opening partly covered by a thin brass plate, formed the shutter. This plate was

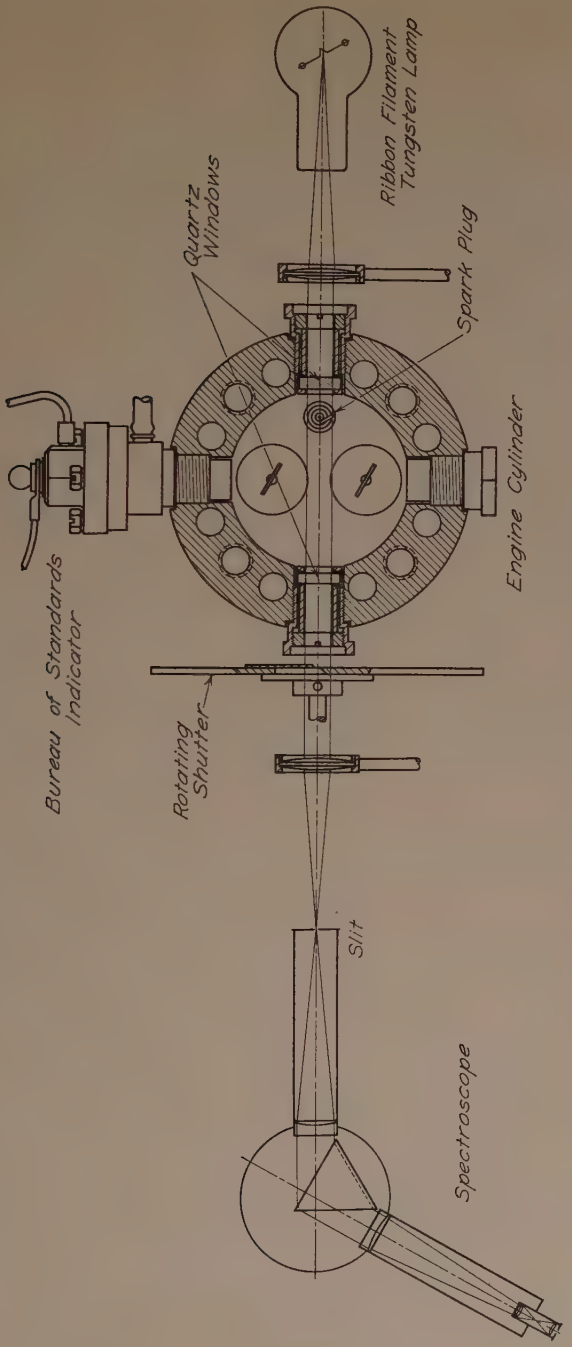


FIG. 5. DIAGRAM OF OPTICAL SYSTEM

held in place by screws passing through circular slots so that it could be shifted to change the angular opening. The drive for the shutter was taken from the camshaft through a train of bevel gears which included a phase-changing gear, by means of which the timing of the shutter relative to the crankshaft could be changed. Rotating the housing of the phase-changing gear through a given angle moved the shutter through twice this angle relative to the crankshaft. The housing was rotated by a worm gear, and a counter, driven from the shaft of the worm, indicated the phase relation between the shutter and crankshaft. As it was difficult to eliminate lash entirely from such a gear train, which included five pairs of bevel gears, reasonable care was exercised in setting the gears, and the actual phase difference due to this lash was measured. This was done by placing a spark plug in the window opening on the opposite side of the cylinder and determining the apparent spark timing by changing the phase of the shutter until the spark was first observed, the line of sight being determined by two screens with narrow openings. The engine was operated at normal speed and the actual spark timing was found from the spark indicator on the crankshaft. The difference between the actual and the apparent timing was taken to be the phase error of the shutter due to lash. Except for a narrow range of about $\pm 5$ degrees from top center, the phase error was constant in amount and direction, so that it was only necessary to subtract this amount from the counter reading to obtain the actual timing of the shutter.

8. *Pressure-Measuring Apparatus.*—A Bureau of Standards balanced-diaphragm indicator was used in measuring the gas pressure in the engine cylinder, the pressure element being screwed into one of the holes in the cylinder wall at 90 degrees from the windows. At first considerable difficulty was experienced with the indicator due to diaphragm failure; the regular plated diaphragms quickly corroded even though they were frequently removed and cleaned. This was overcome by using stainless steel diaphrams 0.006 in. thick.* The contact timer for the indicator was driven from the camshaft through the phase-changing gear which drove the stroboscope shutter. Thus, after the timer and shutter were set in phase, they could simultaneously be shifted to any point in the cycle by adjusting the phase-changing gear without entailing any change in their relative timing. The phase difference with respect to the crankshaft was found by measuring the spark timing simultaneously with the indicator incor-

*The steel, an 8 per cent nickel, 18 per cent chrome alloy, was furnished by the Cold Metal Process Company, Youngstown, Ohio.

porated in the timer and the one mounted on the crankshaft, the difference between two corresponding readings being the phase error caused by lash in the driving gear. Corrections could then be made as previously described for the phase error of the rotating shutter.

Instantaneous pressures in the inlet manifold were also measured with the same type of indicator. A piezometer ring around the inlet manifold close to the cylinder head served as a pressure tap to which the indicator was connected with a short ⅛-inch pipe nipple. Pressure measurements were taken during the induction period, starting some time before the inlet valve opened and continuing until after it had closed.

## IV. Experimental Procedure

9. *Method of Conducting Tests.*—Before starting the engine for a test, the rotating shutter was removed and the spectroscope carefully aligned with the rest of the optical system so that the same part of the image of the filament V always fell on the slit. The spectroscope was then clamped in place and, after the shutter had been replaced, the engine was started. No sodium was introduced into the cylinder until the engine had been running for a sufficient time to heat the cylinder and manifold jacket approximately to operating temperatures. Any unvaporized portions of the NaOH solution that reached the combustion chamber collected on the walls and windows and quickly pitted the latter. It was necessary, therefore, to avoid introducing any excess of the solution, only enough being aspirated into the manifold to give satisfactory reversal of the spectral line, and this only after the engine was thoroughly warmed up. The carburetor needle valve was next adjusted to give the desired air–fuel ratio, the fuel and air consumption being measured only when the engine operating conditions had become constant. A series of flame temperature measurements were then made, starting with the piston at its top center position and continuing at intervals of ten degrees of crank rotation to a point in the operating cycle well past that at which the maximum temperature was observed. No temperatures were measured before the piston reached top center, because, with the exception of those tests in which the mixture was very rich, the burning had not progressed sufficiently before this point in the cycle to produce conditions within the cylinder giving definite reversal of the spectral line. In determining the balance point between the bright and reversed lines it was necessary to select an average condition representative of a large number of separate cycles. For the

temperature was found to vary noticeably from one cycle to another, and so the lamp intensity was adjusted until the bright and reversed lines each appeared with about equal frequency. When the balance point had been found the current through the lamp was measured by means of the potentiometer and recorded. As soon as the traverse around the cycle was completed the air and fuel consumptions were measured again, and a second series of temperature measurements taken. These observations were also made at ten-degree crank intervals, starting midway between the first two readings of the previous set, thus giving temperatures at five-degree intervals as the final result. When this series was completed the air and fuel consumption were measured once more and the engine stopped. While a test was in progress the jacket and manifold temperatures and the speed were frequently checked, and adjustments were made when necessary, but no change was made in the carburetor setting after the initial adjustment.

10. *Lamp Calibration.*—When flame temperatures were being measured, the intensity of the radiation from the sodium vapor and that of the radiation from the lamp filament having the same wavelength were balanced in the region within the engine cylinder. Therefore, the brightness temperature of the lamp radiation inside the cylinder was determined for the required range of filament currents by removing the rotating shutter and the left cylinder window, see Fig. 5, and calibrating the lamp as described in Section 7. Any absorption or reflection due to the shutter and this window, or to the optical parts of the spectroscope, reduced the intensity of the radiation from the lamp and the sodium vapor by exactly the same amount, and hence they were removed from the optical system during calibration.

In order to determine the reduction in intensity of the lamp radiation, caused by the deposit on the inner surface of the right cylinder window, the lamp was calibrated at the end of each test. With lean mixtures the decrease in intensity was negligible, but with rich mixtures the brightness temperature was sometimes lowered by as much as 100 deg. F. In any case, most of the deposit occurred while the engine was warming up, or while operating conditions were being adjusted, so that a calibration at the end of a test was found to be representative of conditions during the period in which temperature measurements were being made.

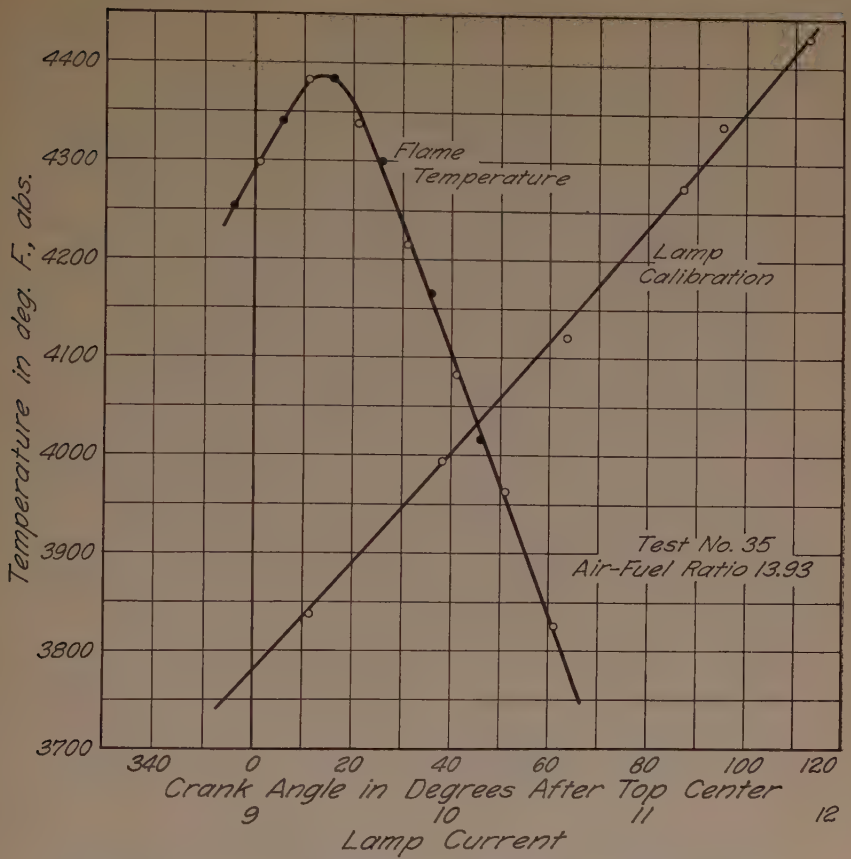


FIG. 6. SAMPLE FLAME TEMPERATURE AND LAMP CALIBRATION CURVES

## V. TEST RESULTS

11. *General Statement.* — The line-reversal method, while well established as a means of measuring the temperature of stationary flames, has not been used extensively in measuring rapidly changing temperatures, such as those which occur in a bomb or an engine cylinder. Accordingly, it was essential, in applying the method to the measurement of flame temperatures in an engine, to repeat the experiment many times to determine what accuracy might be expected. As already pointed out, every effort was made to keep all conditions

TABLE 1

SAMPLE TEST DATA

Data from Test No. 35

Air-Fuel Ratio 13.93 Observers: RFP and AEH

Spark Advance 36 deg. Date: January 25, 1933

Flame Temperature Measurements

| Phase Gear Counter Reading | Crank Angle deg. | Lamp Current* | Temp. deg.F.abs. |
|---|---|---|---|
| 920 | 71 | 8.748 | .... |
| 930 | 61 | 9.238 | 3825 |
| 940 | 51 | 9.715 | 3963 |
| 950 | 41 | 10.353 | 4083 |
| 960 | 31 | 10.922 | 4215 |
| 970 | 21 | 11.440 | 4337 |
| 980 | 11 | 11.634 | 4383 |
| 990 | 1 | 11.277 | 4298 |
| 945 | 46 | 10.068 | 4016 |
| 955 | 36 | 10.706 | 4165 |
| 965 | 26 | 11.285 | 4300 |
| 975 | 16 | 11.628 | 4382 |
| 985 | 6 | 11.452 | 4340 |
| 995 | 356 | 11.083 | 4252 |

Calibration of Lamp (W-79)

| Lamp Current* | Pyrometer Screen | Pyrometer Current ma. | Temp. deg.F.abs. |
|---|---|---|---|
| 9.291 | Ex. High | 406 | 3837 |
| 9.956 | Ex. High | 417 | 3993 |
| 10.583 | Ex. High | 427 | 4120 |
| 11.177 | Ex. High | 438 | 4274 |
| 11.378 | Ex. High | 443 | 4339 |
| 11.823 | Ex. High | 450 | 4430 |

*Potential drop across shunt in light circuit in milli-volts.

uniform during any given test. The results of one such test, together with the lamp calibration, made immediately after its completion, are given in Table 1 and plotted in Fig. 6. It may be noted that the deviation of the individual points from a smooth curve drawn to represent the observed changes is surprisingly small; but it was found that the agreement between comparable sets of observations taken at different times was not as close as might be expected from the consistency of the results obtained during a single run. In searching for the cause of comparatively large variations between runs, variations in atmospheric humidity were checked, but such changes as occurred were found to have little or no effect on the measured results. The installation of a more accurate spark indicator increased the precision with which analogous runs would check, and for this

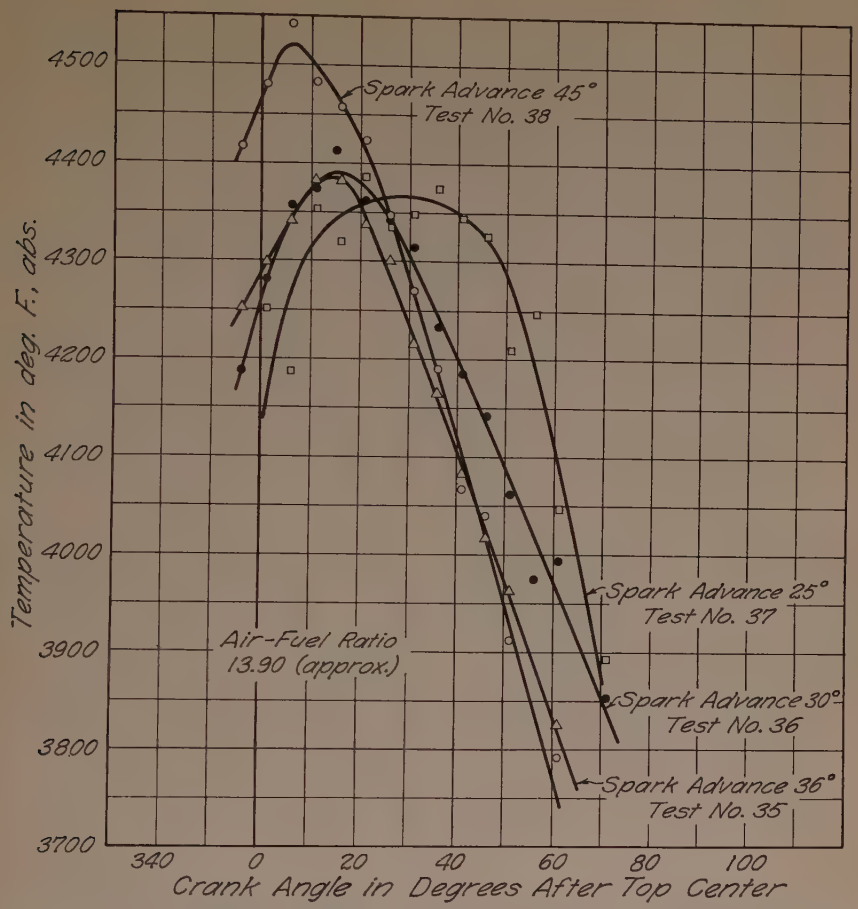


FIG. 7. EFFECT OF SPARK TIMING ON TEMPERATURE—CONSTANT AIR-FUEL RATIO

reason a series of runs were made with three settings of the carburetor giving different air–fuel mixtures, and with different spark settings for each mixture. These, together with a large number of runs made with the 36-deg. spark setting and different air–fuel ratios, constitute the main experimental results included in this bulletin.

12. *Effect of Spark Timing.*—The results of the experiments on spark timing are summarized in the curves of Figs. 7 and 8. The curves of Fig. 7 show how the temperatures changed with the spark timing, the air–fuel ratio being kept as nearly constant as possible. The phase in the cycle at which the maximum temperature occurs is

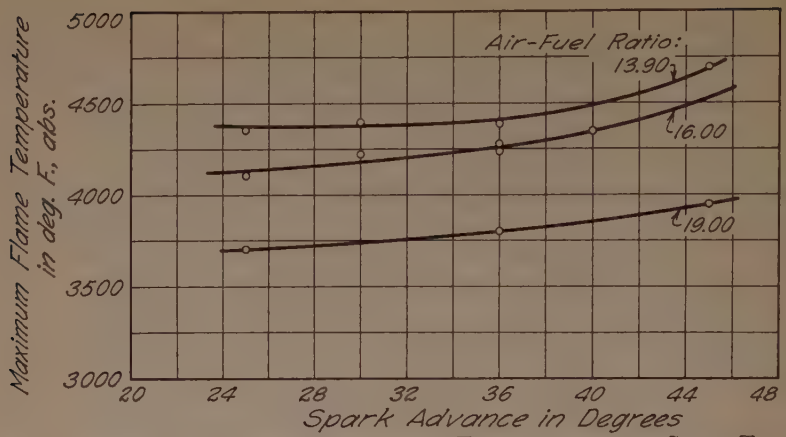


FIG. 8. VARIATION OF MAXIMUM COMBUSTION TEMPERATURE WITH SPARK TIMING

changed, as well as the shape of the temperature curve, by shifts in spark timing. To find the correct position of the spark for each fuel mixture would involve a separate study that was not attempted except for approximately the theoretical air–fuel ratio. As indicated in Section 6, the 36-degree position for the spark timing was determined as the one which gave the maximum power for this ratio.

In Fig. 8 the curves show how the maximum temperatures changed as the spark timing was shifted with three different fuel mixtures. Though the data are incomplete, they, nevertheless, indicate quite clearly the influence of spark timing on the maximum temperatures. It will be noted that beyond the 36-degree position maximum temperatures change more rapidly with increasing spark advance, especially for the richer mixtures. But these extreme spark positions are of little practical importance, for with such spark advance engine operation becomes increasingly rough and irregular, while temperature measurements become very uncertain, particularly for the leaner mixtures.

13. *Effect of Varying Air–Fuel Ratio.*—After the installation of the more exact spark timing device and the development of a satisfactory comparison lamp, the agreement between comparable runs was much improved. Altogether some forty-nine runs were made, and when the technique of the operation was perfected these could be checked readily within a range of $\pm 50$ deg. F. No evidence was observed of self-reversal in the sodium lines, and since the resolution of the spectroscope was sufficient to show the two D-lines well resolved

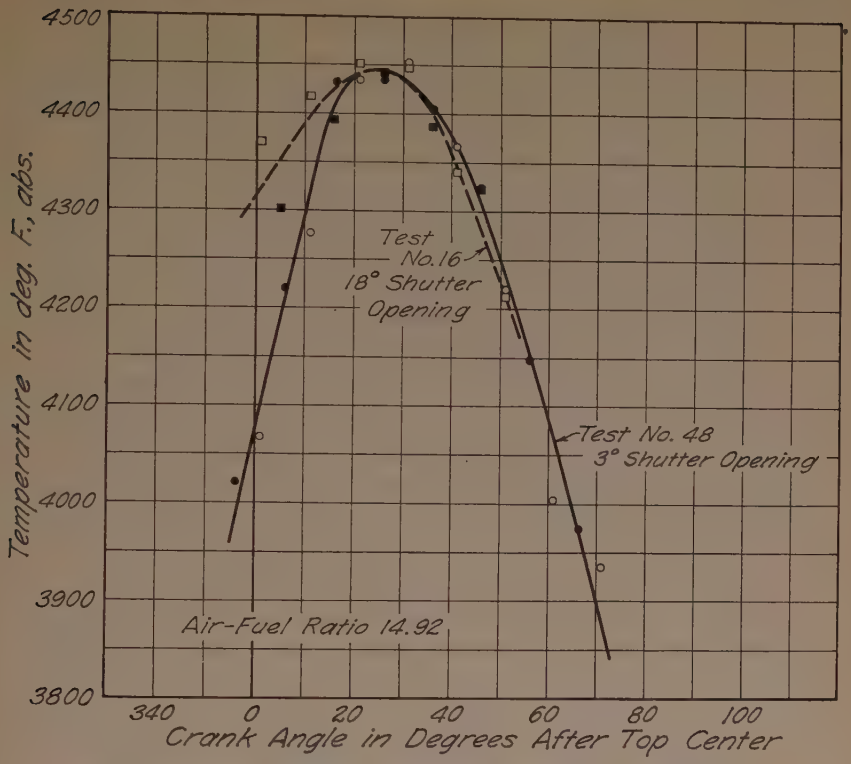


FIG. 9. EFFECT OF VARIATION OF SHUTTER OPENING ON TEMPERATURE MEASUREMENTS

it was concluded that the layer of cooler gases in contact with the cylinder walls was too thin to affect the results appreciably. Any temperature measured was, of course, an average taken along a diameter of the cylinder, and was averaged also over a sequence of many cycles. The method very definitely showed the variations in temperature from cycle to cycle, and estimates indicate that it could be used to detect temperature fluctuations of approximately 10 deg. F. at the temperatures measured, provided the temperature remained constant while a balance was being observed. To obtain sufficient visibility a finite shutter opening was used, most of the observations being made with light entering the spectroscope during about 18 deg. of crank-angle. However, runs made with the light entering the spectroscope during but 3 deg. of crank angle gave results for maximum temperatures not measurably different from those made with the larger angle. In Fig. 9 two curves are drawn from runs with essentially identical air–

fuel ratios, one made with a 3-degree interval and the other with the 18-degree interval. With proper corrections made for differences in the optical setup, the maxima of the two curves are seen to agree within the limit of experimental error, though the one made with the wider interval is noticeably broader. This indicates that temperature at the maximum remains constant over an appreciable part of the cycle. The flicker accompanying observations with the 3-degree timing was more noticeable, but the variations in the temperatures from cycle to cycle in all the measurements were considerably greater than the precision of the setting.

Complete measurements of gas pressures for six widely differing air–fuel ratios had been made with the engine used in this study by one of the authors in connection with a previous study of radiation measurement.* Pressure measurements were, therefore, not made for each run in this investigation, only two being made as checks. One of the curves taken from the previous study is included here in Fig. 10. This shows also the data from which the volume at the start of compression is determined. The data for the temperatures during part of the cycle for the same air–fuel ratio, but taken by the line-reversal method, are also shown here. In another run, the data for which are not included, it was found that the line-reversal method could be used to measure the temperature in the engine well after the exhaust valve opened and even beyond the bottom center position. The temperature seems to reach maximum values before the pressure, but during the later portions of the cycle the pressure and temperature curves parallel each other quite closely. Most of the runs were made without pressure measurements, and with only enough readings of temperature to completely determine the maximum. The set of curves in Fig. 11 were selected as good examples of the experimental data and as illustrating also the manner in which the temperature varies as the air–fuel ratio is changed. The curves with broken lines were drawn from data taken with fuel mixtures richer than the theoretical, while the full line curves were drawn from data obtained with mixtures leaner than this. The trend toward and away from a maximum temperature is very conspicuous. Observations were made over as wide a range of air–fuel mixtures as could be used, still retaining reasonably smooth engine operation. The highest temperature was obtained with an air–fuel ratio of 13.9 to 1, which corresponds to a mixture containing 91 per cent of the theoretical amount of air.

*Loc. cit., p. 5.

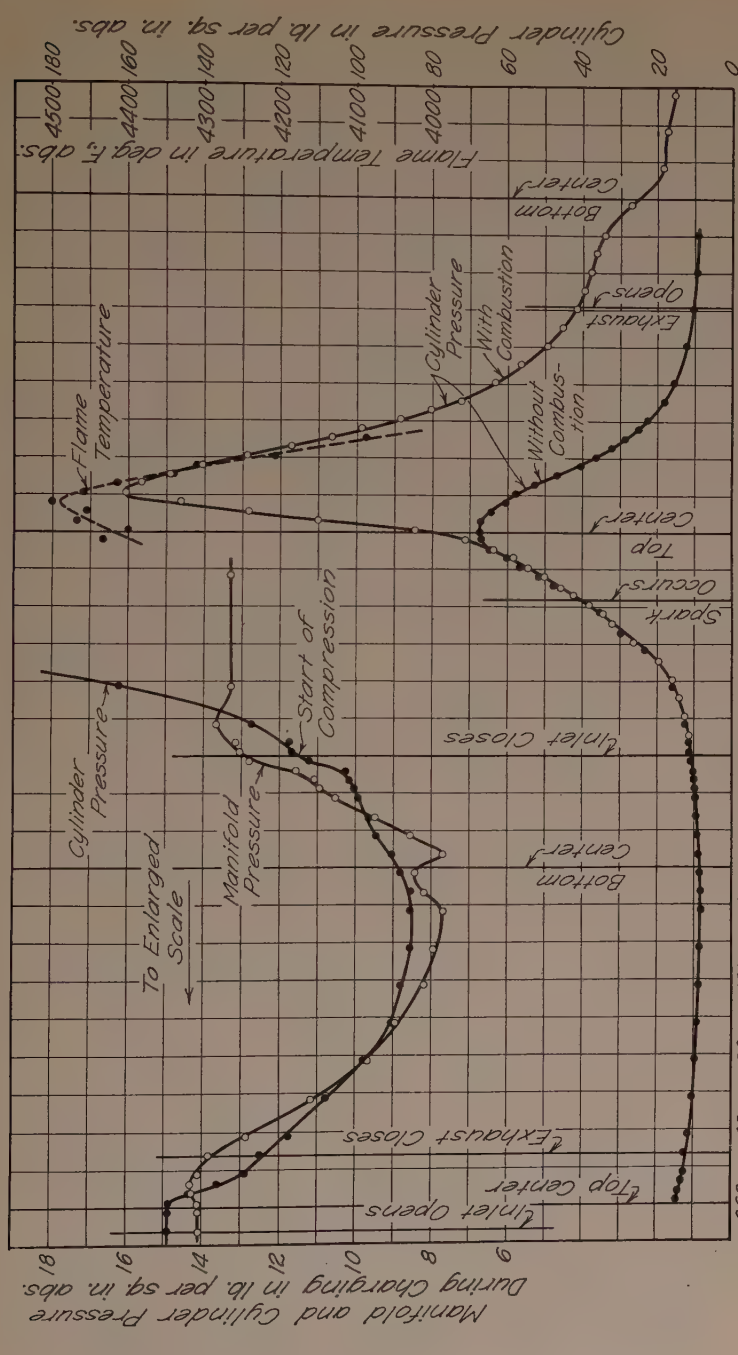


FIG. 10. CYLINDER AND MANIFOLD PRESSURES
Air-Fuel Ratio 13.7:1

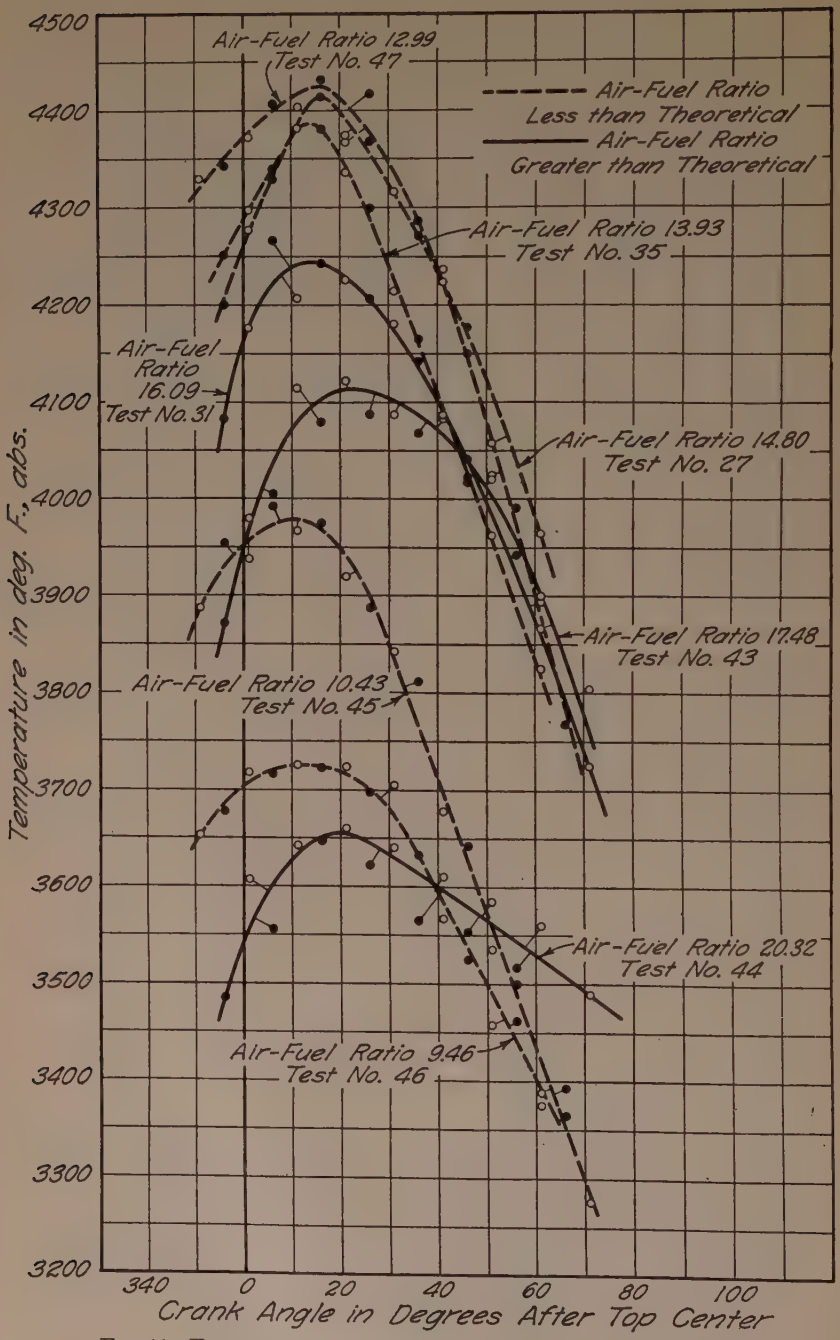


FIG. 11. TEMPERATURE CURVES WITH VARYING AIR-FUEL RATIOS

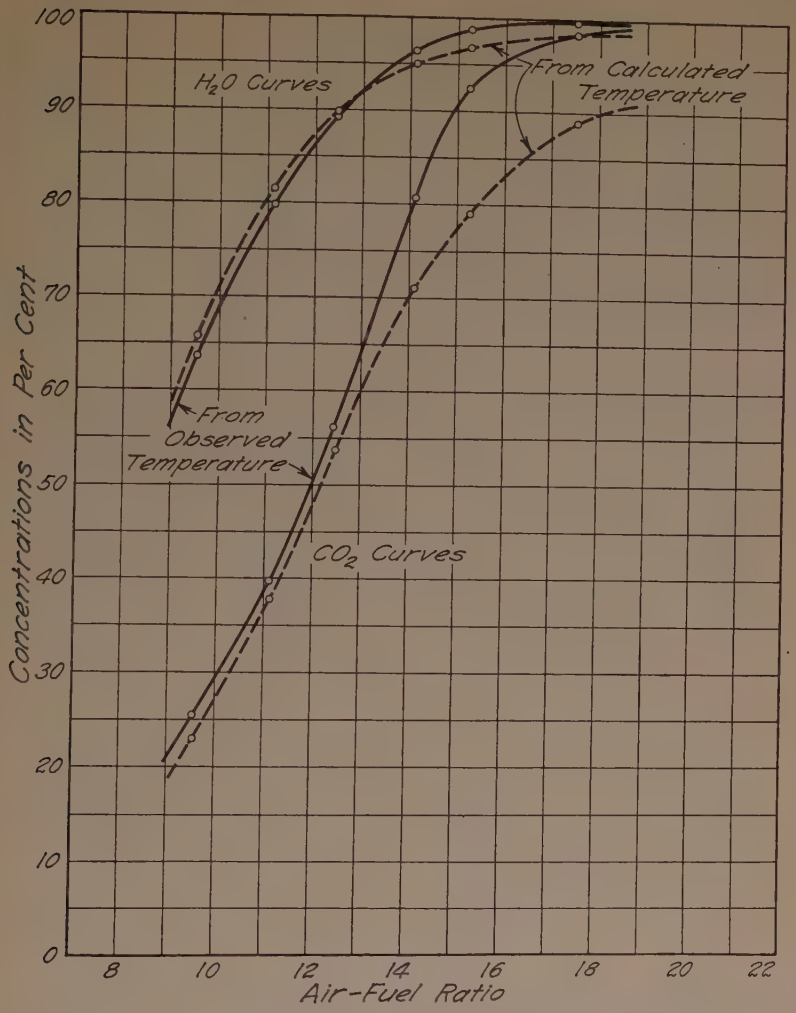


FIG. 12. CONCENTRATION OF $H_2O$ AND $CO_2$ AT MAXIMUM OBSERVED AND CALCULATED TEMPERATURES

14. *Concentrations of Combustion Products.*—When the instantaneous temperatures of the gaseous mixture inside the engine cylinder are known, the corresponding concentrations of the components can be computed if correct equilibrium data are available. Thus, the concentrations of $H_2O$ and $CO_2$ were computed for the maximum observed temperature at six representative air–fuel ratios from the equilibrium data for the three reactions,

$$H_2 + \tfrac{1}{2}O_2 \rightleftharpoons H_2O$$
$$CO + \tfrac{1}{2}O_2 \rightleftharpoons CO_2$$

and
$$H_2 + CO_2 \rightleftharpoons H_2O + CO$$

assuming that combustion occurs at constant volume. The results of these calculations are shown graphically in Fig. 12, where the percentages of the concentration for complete combustion are plotted against air–fuel ratio. In a similar manner concentrations were computed for the maximum temperatures determined by theoretical considerations and these results are also shown in Fig. 12. The data from which the curves in this figure were plotted are given in Tables 2 and 4.

It may be noted that the concentrations found for the observed and calculated temperatures are in reasonable agreement up to an air–fuel ratio of 12.5 to 1. As the air–fuel ratio increases beyond this value, both the $H_2O$ and $CO_2$ concentrations for the observed temperatures increase more rapidly than the corresponding concentrations for the calculated temperatures. This is due to an increasing difference between the observed and calculated temperatures, the former being lower and hence allowing more complete combustion.

## VI. Comparison with Theoretical Calculations

15. *Method of Calculating Temperatures.* — By means of the method developed by Goodenough and Felbeck,* the maximum flame temperature during combustion in an engine may be calculated. This has been done by Goodenough and Baker† for a wide range of compression pressures and air–fuel ratios. By interpolating between their results, the maximum temperatures could be found for a compression ratio of 3.86 to 1, which was the actual ratio from the closing of the inlet valve in all the present tests. This was done for a number of air–fuel ratios, and the results are represented graphically by the upper full line in Fig. 13. The broken line in the same figure represents calculated temperatures found by slightly modifying the method mentioned.

In considering the Otto cycle with insufficient air, Goodenough and Baker assumed that the residual gas mixture in the clearance volume, when the fresh charge enters, contains no $O_2$.‡ This is not strictly

*"An Investigation of the Maximum Temperatures and Pressures Attainable in the Combustion of Gaseous and Liquid Fuels," Univ. of Ill. Eng. Exp. Sta. Bul. No. 139. (1924).
†Loc. cit., p. 5.
‡Page 28, loc. cit., p. 5.

TABLE 2

SUMMARY OF CALCULATED RESULTS

| Air–Fuel Ratio by Weight | Percentage of Theor. Air | $T_1$ deg. F. abs. | $P_1$ lb. per sq. in. abs. | $m_1$ mols. | $n_1$ | $n_2$ | $n_2'$ | $T_2$ deg. F. abs. | $P_2$ lb. per sq. in. abs. | $T_3$ deg. F. abs. | $P_3$ lb. per sq. in. abs. | $x_3$ per cent | $y_3$ per cent |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 9.55 | 63.0 | 728.6 | 11.5 | 41.81 | 8.535 | 9.602 | 4.134 | 1092 | 66.58 | 4402 | 320.5 | 22.89 | 65.75 |
| 11.14 | 73.5 | 746.0 | 11.5 | 48.18 | 8.496 | 9.559 | 5.510 | 1130 | 67.32 | 4753 | 323.1 | 37.88 | 81.44 |
| 12.44 | 82.1 | 759.0 | 11.5 | 53.42 | 8.474 | 9.533 | 6.634 | 1150 | 67.28 | 4956 | 321.6 | 53.75 | 89.86 |
| 14.10 | 93.1 | 773.9 | 11.5 | 60.15 | 8.455 | 9.512 | 8.068 | 1191 | 68.37 | 5046 | 313.3 | 71.08 | 95.06 |
| 15.27 | 100.8 | 779.8 | 11.5 | 64.98 | 8.451 | 9.507 | 9.082 | 1206 | 68.73 | 5000 | 304.7 | 79.01 | 96.68 |
| 17.52 | 115.6 | 780.0 | 11.5 | 74.81 | 8.456 | 9.513 | 11.059 | 1219 | 69.42 | 4834 | 290.3 | 88.72 | 98.30 |

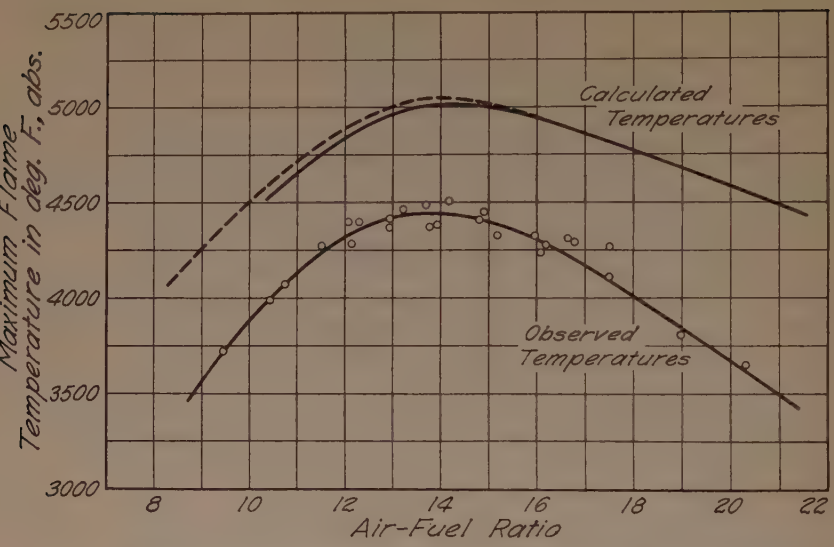


FIG. 13. OBSERVED AND CALCULATED MAXIMUM COMBUSTION TEMPERATURES

true because at the temperature of this residual gas (2800 to 3400 deg. F. abs.) some dissociation must always occur. The gas mixture which would be in equilibrium under the existing conditions was found by evaluating the two equilibrium equations

$$\log K_{p(CO)} + \tfrac{1}{2} \log T = \log x - \log (1 - x) + \tfrac{1}{2} \log \frac{N_i T_2}{P_2} - \tfrac{1}{2} \log n'_e \qquad (12)$$

and

$$K_{p(w.g.)} = \frac{y\,(1 - x)}{x\,(1 - y)} \qquad (13)$$

for the temperature $T$ of the residual gases, and solving for $x$ and $y$. With this exception the method of calculation was the same as that used by Goodenough and Baker. However, instead of the initial conditions assumed by them, such as the temperature and pressure of the charge in the inlet manifold when the inlet valve opens, actual experimental values were used. The temperatures were found by means of the thermometer in the manifold, while the pressures were obtained

TABLE 3

SUMMARY OF TEST RESULTS

Compression Ratio—3.86:1 Spark Advance—36 deg.

| Test No. | Date of Test | Air Pressure at Meter in. of Hg. abs. | Air Temperature at Meter deg. F. | Moisture Content lb. $H_2O$ vapor per 100 lb. of air | Inlet Temperature deg. F. | Air–Fuel Ratio* | Maximum Temperature deg.F.abs. | Crank Position at Max. Temp., deg. past top center |
|---|---|---|---|---|---|---|---|---|
| 8 | 10- 6-32 | 29.34 | 87.5 | 0.386 | 115 | 15.20 | 4330 | .... |
| 9 | 10-10-32 | 29.24 | 89.0 | 0.520 | 118 | 12.15 | 4315 | .... |
| 13 | 10-31-32 | 29.25 | 77.7 | 0.502 | 120 | 12.09 | 4400 | 11.0 |
| 15 | 11- 7-32 | 29.17 | 90.0 | 0.736 | 122 | 12.30 | 4400 | 13.5 |
| 16 | 11- 7-32 | 29.05 | 93.5 | 0.997 | 122 | 14.92 | 4450 | 15.0 |
| 17 | 11-14-32 | 29.12 | 86.2 | 0.342 | 120 | 17.51 | 4270 | 14.0 |
| 18 | 11-14-32 | 29.09 | 85.0 | 0.535 | 119 | 10.76 | 4075 | 7.0 |
| 19 | 11-14-32 | 29.10 | 87.0 | 0.650 | 120 | 11.53 | 4275 | 9.0 |
| 22 | 11-21-32 | 29.66 | 84.2 | 0.133 | 120 | 13.70 | 4490 | 15.0 |
| 23 | 11-21-32 | 29.67 | 84.5 | 0.181 | 116 | 13.23 | 4470 | 11.0 |
| 24 | 12- 5-32 | 29.18 | 90.8 | 0.334 | 122 | 13.79 | 4370 | 10.0 |
| 26 | 12-22-32 | 29.67 | 83.0 | 0.378 | 122 | 16.19 | 4280 | 18.0 |
| 27 | 1- 9-33 | 29.35 | 87.0 | 0.306 | 120 | 14.80 | 4412 | 16.0 |
| 28 | 1- 9-33 | 29.28 | 88.5 | 0.317 | 125 | 15.97 | 4325 | 16.0 |
| 29 | 1-16-33 | 29.20 | 85.7 | 0.690 | 125 | 16.79 | 4295 | 18.0 |
| 30 | 1-16-33 | 29.11 | 88.5 | 0.621 | 125 | 14.18 | 4505 | 11.0 |
| 31 | 1-23-33 | 29.18 | 87.2 | 0.421 | 125 | 16.09 | 4242 | 17.0 |
| 35 | 1-25-33 | 28.95 | 82.5 | 0.564 | 125 | 13.93 | 4385 | 13.5 |
| 39 | 1-28-33 | 29.47 | 84.0 | 0.257 | 124 | 18.99 | 3805 | 25.0 |
| 42 | 2-13-33 | 29.29 | 81.2 | 0.199 | 123 | 16.64 | 4318 | 21.0 |
| 43 | 2-13-33 | 29.15 | 83.8 | 0.380 | 125 | 17.48 | 4112 | 21.0 |
| 44 | 2-13-33 | 29.13 | 83.7 | 0.427 | 126 | 20.32 | 3655 | 19.0 |
| 45 | 2-14-33 | 29.45 | 82.0 | 0.297 | 119 | 10.43 | 3980 | 11.0 |
| 46 | 2-14-33 | 29.46 | 82.7 | 0.272 | 117 | 9.46 | 3725 | 13.0 |
| 47 | 2-16-33 | 29.03 | 84.8 | 0.309 | 121 | 12.99 | 4425 | 14.0 |
| 48 | 4-24-33 | 29.05 | 85.3 | 0.449 | 120 | 14.91 | 4445 | 21.0 |
| 49 | 4-24-33 | 28.93 | 85.7 | 0.383 | 121 | 12.96 | 4370 | 12.0 |

*Calculated for dry air.

from curves of manifold pressure such as that shown in Fig. 10. From these same curves the volume at the beginning of compression, and hence the actual compression ratio, could also be found. A summary of these calculated results is given in Table 2. The net effect of these modifications was to leave the temperatures for lean mixtures unchanged, and to raise those for rich mixtures by 50 to 75 deg. F. as indicated by the broken line portion of the upper curve in Fig. 13.

16. *Comparison of Calculated and Observed Temperatures.*—The observed maximum temperatures, given in Table 3, were also plotted against air–fuel ratios, and are represented by the lower curve in Fig. 13. The general shape of this curve is the same as that of the curve for calculated values, but the latter is from 600 to 1000 degrees higher. The smallest difference occurs with an air–fuel ratio of 14 to 1, when both calculated and observed temperatures are highest.

This difference is undoubtedly due to a number of factors, the more obvious ones being the following:

(a) Effect of heat loss and work done up to position of maximum temperature
(b) Effect of NaOH solution
(c) Effect of humidity
(d) Effect of finite shutter opening
(e) Effect of fogging of the cylinder window
(f) Effect of precision of the line-reversal method
(g) Effect of chemical equilibrium conditions

**(a) Effect of heat loss and work done up to position of maximum temperature**

In calculating the temperatures, the assumption was made that combustion takes place adiabatically and at constant volume. Under such conditions the energy equation

$$\Delta Q = \Delta U + \Delta W \qquad (14)$$

becomes

$$H_m - n_1 (1 - x) H_{co} - n_2 (1 - y) H_{H_2} = U_3 - U_2 \qquad (15)$$

using the notation of Bulletin No. 160.* In the actual engine, neither of these conditions is satisfied since combustion continues over a finite interval during which heat loss must occur to the cylinder walls, and work must be done on the moving piston. Therefore, the energy equation should be

$$H_m - n_1 (1 - x) H_{co} - n_2 (1 - y) H_{H_2} - h = U_3 - U_2 + \Delta W \qquad (16)$$

where $h$ is the heat loss, and $\Delta W$ the work done. For any given test, $\Delta W$ may be found from the area under the pressure-volume curve between top center position and that at which maximum temperature occurs. Now if it is assumed that the difference between calculated and observed temperatures is due entirely to this work of expansion

*See Equation (21), page 21, loc. cit. $H_m$ is the lower heat of combustion at constant volume for the combustible mixture at the end of compression. This mixture contains the gasoline of the fresh charge together with the $H_2$ and CO of the residual gases left in the cylinder at the end of exhaust. The values given by Goodenough and Felbeck were used in calculating all heats of combustion. For gasoline they assume the higher heat of combustion at constant pressure to be 20 000 B.t.u. per lb., while the gasoline used in these tests had a heat of combustion of 20 400 B.t.u. per lb. The use of this latter value would have resulted in higher calculated flame temperatures; for any increase in $H_m$ in Equation (15) above would necessitate an increase in $U_3$ or an increase in $T_3$, the temperature at the end of combustion.

TABLE 4

RESULTS CALCULATED FROM OBSERVED TEMPERATURE

| Air–Fuel Ratio by Weight | Observed Temp. deg. F. abs. | $x_3$ per cent | $y_3$ per cent | Energy Calculations on $m_1$ mols of Combustion | | | $\Delta W$ per cent $H_m$ | Apparent Heat Loss per cent $H_m$ |
|---|---|---|---|---|---|---|---|---|
| | | | | $H_m$ $10^4$ B.t.u. | $\Delta W$ $10^4$ B.t.u. | $h$ $10^4$ B.t.u. | | |
| 9.55 | 3754 | 25.43 | 63.50 | 222.2 | 0.795 | 24.69 | 0.4 | 11.1 |
| 11.14 | 4155 | 39.89 | 79.81 | 220.0 | 0.896 | 27.33 | 0.4 | 12.4 |
| 12.44 | 4378 | 56.16 | 89.16 | 218.4 | 1.261 | 30.81 | 0.6 | 14.1 |
| 14.10 | 4435 | 80.69 | 96.47 | 216.6 | 1.736 | 44.82 | 0.8 | 20.7 |
| 15.27 | 4384 | 92.49 | 98.75 | 215.5 | 2.371 | 50.90 | 1.1 | 23.6 |
| 17.52 | 4115 | 98.29 | 99.70 | 215.4 | 2.626 | 56.55 | 1.2 | 26.3 |

and heat loss, the latter quantity may also be determined. When an observed temperature has been substituted in the equilibrium equations (12) and (13), the corresponding values of $x$ and $y$ may be calculated, and $h$ found from Equation (16) after substituting these values of $x$, $y$, and $\Delta W$. Table 4 contains the results of such calculations for six different air–fuel ratios. The point in the cycle at which maximum temperature occurs is somewhat indefinite, as may be seen from Fig. 14, in which is shown the crank-angle at maximum temperature plotted against air–fuel ratio for a large number of tests. The positions of maximum temperature, used in calculating $\Delta W$, were found from the curve in this figure. $\Delta W$ and $h$, expressed as percentages of the heat of combustion $H_m$, were plotted against air–fuel ratios, as shown in Fig. 15. It is evident from this graph that the work of expansion $\Delta W$ is far too small to account for much of the difference between calculated and observed maximum temperatures. Moreover, the heat loss would have to attain unreasonable values if it alone were to account for the temperature difference. For, as may be seen in Fig. 15, this would require losses of from 11 per cent at rich mixtures up to 26 per cent at lean mixtures; while experimental evidence would apparently indicate that the heat loss during combustion up to the point of maximum temperature should not exceed 5 per cent.*

### (b) Effect of NaOH Solution

The amount of NaOH solution introduced into the engine cylinder was so small that no very appreciable lowering of the temperature could be expected unless it had some catalytic effect on combustion. Jones, Lewis, Friauf, and Perrott have determined the effect of NaCl

*"Some Properties of the Working Fluid of Gas Engines," W. T. David. Engineering, Vol. 113, p. 281. (1922)

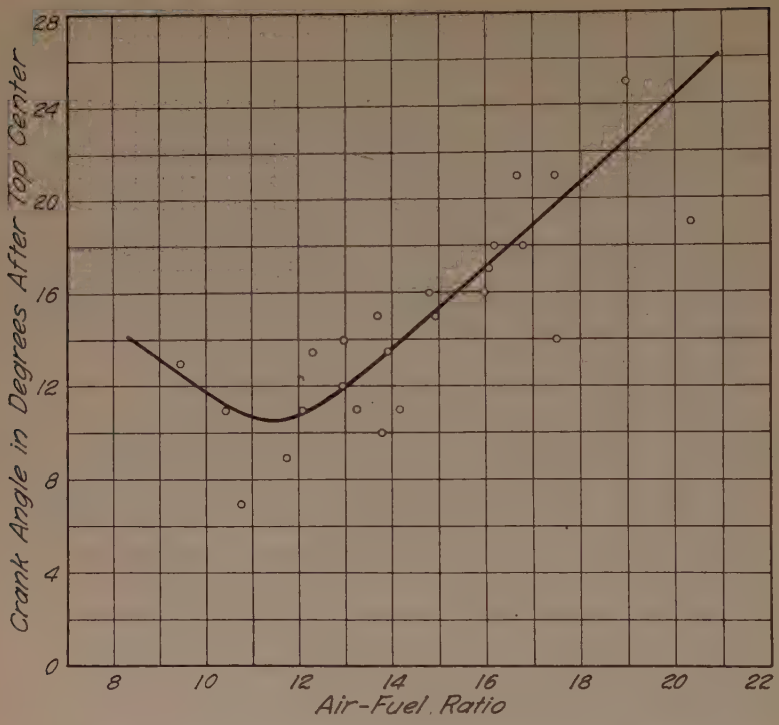


FIG. 14. CRANK ANGLE AT MAXIMUM COMBUSTION TEMPERATURE

solution on the temperature of a stationary flame in connection with their investigation of flame temperatures by line reversal.* From measurements of the brightness temperature of a platinum ribbon heated in the flame, they estimated that with a flame temperature of 3875 deg. F. abs., the introduction of enough solution to give definite reversal reduces the flame temperature about 27 deg. F.

### (c) Effect of humidity

Goodenough and Felbeck have found that water vapor added to the combustible mixture lowers the calculated flame temperatures.† However, this effect becomes appreciable only when the amount of water added is greater than 10 per cent of the fuel burned, the decrease in temperature for this particular mixture being about 50 deg. F. From the humidity data included in Table 3 it may be seen that

*Loc. cit., p. 8.
†Loc. cit., p. 34.

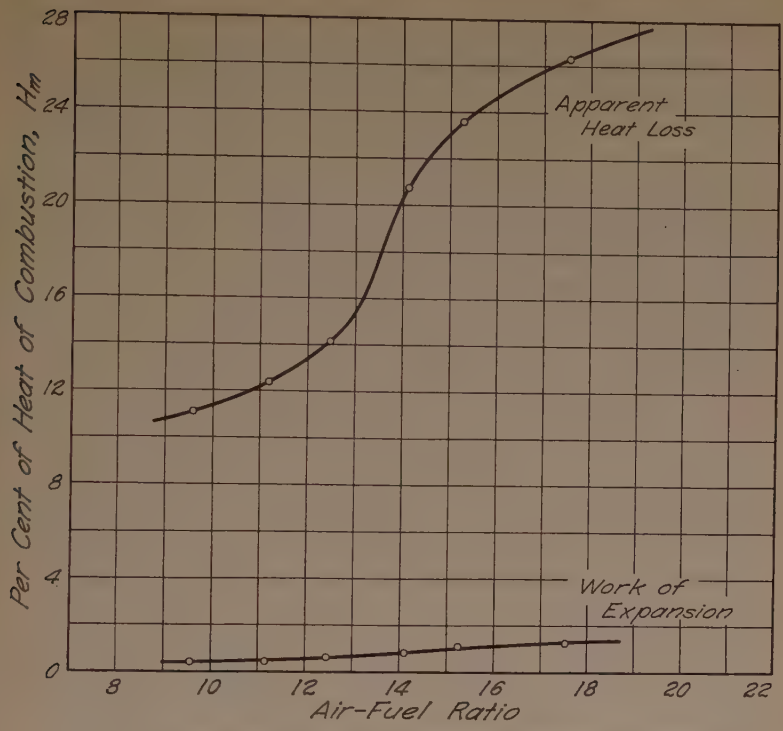


FIG. 15. APPARENT LOSSES AT MAXIMUM OBSERVED TEMPERATURES

in only one instance did the moisture content of the air approach a value of one pound per hundred pounds of air. In this extreme case the water vapor would still be less than 10 per cent of the fuel with even the richest mixture that could be burned. It would, therefore, be reasonable to expect that the agreement between calculated and observed temperatures would not be measurably improved by correcting the calculations for the presence of water vapor in the combustible. Jones* has calculated the temperatures of a stationary flame, assuming first dry and then saturated air. In the former case the temperature was found to be about 90 deg. F. higher than in the latter.

### (d) Effect of finite shutter opening

As has already been explained, the observed temperatures are average values over the finite time interval during which the shutter

*Loc. cit., p. 8.

was open. Since the opening was about 18 deg. and readings were taken at 5 deg. intervals, the effect would be that of a running average in which the temperature at any given crank position would be included in at least four observations. In spite of this, it might be possible to obtain observed maximum temperatures considerably lower than the actual temperatures, due to this averaging process, if the time during which the latter persisted were extremely short. Decreasing the shutter opening to 3 deg. of crank travel showed no temperature increase, so that it may safely be concluded that the finite shutter opening would not account for the difference between observed and calculated maximum temperatures.

#### (e) Effect of fogging of the cylinder window

Any deposit which accumulated on the window, between the time when observations were being made and that when the lamp was calibrated, would tend to lower the brightness of the lamp radiation inside the engine cylinder. Hence, the curve drawn for an actual calibration would be lower than one drawn for a calibration made simultaneously with the flame temperature observations. However, if the window deposit were increasing very rapidly during the time these observations were being made, the apparent temperatures as read from the actual calibration curve should be consistently higher for the second traverse than for the first. That this is not the case, may be seen from the curves in Fig. 11, where the circles are used to indicate the first set of readings and the filled circles the second set. Therefore, the fogging of the cylinder window does not furnish a satisfactory explanation of the difference between observed and calculated temperatures.

#### (f) Effect of precision of the line-reversal method

Several investigators have measured stationary flame temperatures, both by the line-reversal method and by various other independent methods, and have found the results to be in satisfactory agreement. One such method consists of heating a platinum strip electrically in a vacuum and determining the relation between the brightness temperature and the heating current. The electrically-heated strip is then placed in a flame and a similar relation between temperature and current is found. When these two temperature-current curves are plotted they will intersect, and from the temperature at the point of intersection the flame temperature may be determined. For at this point the strip is dissipating the same amount of energy whether it is in a vacuum or in the flame. When it is in the

former the loss is by radiation alone and, since its temperature is unchanged, this must also be the case in the flame. If there is no direct loss to the flame it and the strip must be in thermal equilibrium and the flame temperature will then be equal to the true temperature of the strip. Loomis and Perrott,* using this method, found the temperature of a natural gas-air flame to be 3672 deg. F. abs. or about 36 deg. F. above the corresponding line-reversal temperature. Griffiths and Awberry,† employing the same method, found temperatures from 11 to 5 deg. F. above the line-reversal temperatures, depending upon the size of the platinum wire which was used. These results certainly indicate that temperatures found by the line-reversal method differ from the actual flame temperatures by amounts which are well within the limits of experimental error.

**(g) Effect of chemical equilibrium conditions**

In discussing the influence of heat loss and work of expansion under (a), it was pointed out that the calculated temperatures are for constant volume adiabatic combustion. Departure from these assumed conditions affects not only the energy equation as already explained, but also the equilibrium equation (8). Since the actual combustion occurs while both pressure and volume are changing, the correct form of this latter equation cannot be readily determined. As a first approximation, it is satisfactory to assume constant volume combustion, for the volume change is much smaller than the pressure change, but a more exact analysis should also include the effect of the changing volume during combustion. In this respect the analysis of the spark-ignition cycle would appear to be quite similar to that of the "mixed" cycle for compression-ignition engines.

Relative volume changes for combustion with various air-fuel ratios may be estimated from the curve in Fig. 14, which shows the angular position of the crankshaft at the time that maximum temperature occurs for different air-fuel ratios. From this curve it is evident that a greater volume change takes place with leaner mixtures than with rich ones, the variation being practically linear for ratios greater than 13 to 1. Since the greatest difference between calculated and observed temperatures was found for the leanest mixture, this would seem to indicate, qualitatively at least, that the departure from constant volume combustion is an important factor in explaining the difference between calculated and measured temperatures. For, assuming that each of the factors discussed under (a), (b), (c), and (f)

*Loc. cit., p. 8.
†Loc. cit., p. 8.

influence the observed results to a maximum extent, the net effect would be a reduction in temperature of some 400 deg. F. If the observed temperatures were increased by this amount they would still be from 200 to 600 deg. F. below the calculated temperatures. And, when it is considered that the combustible is decidedly non-homogeneous, such differences might be reasonably accounted for on the basis of an inadequate analysis of the combustion process.

Lewis, Seman and Jones, using the line-reversal method, found the temperature for a CO-air flame to be 4005 deg. F. abs.* The calculated temperature, considering the effect of variable specific heats and dissociation, was found to be about 270 deg. F. higher than the observed temperature. They concluded that such a difference was too large to be accounted for by radiation loss alone, and were inclined to attribute it to inaccurate specific heat data. For their calculations they used the same specific heat equations as were used in calculating the engine flame temperatures, these being the ones developed by Goodenough and Felbeck.† While new specific heat data would seem to indicate that these equations give inaccurate results at very high temperatures, for the temperature range below 5000 deg. F. abs. the agreement between the equations and the experimental values is still satisfactory. Therefore, although the difference between the calculated and observed temperatures found by Lewis, Seman, and Jones is of same order as those which remain unaccounted for in the case of the engine flame temperatures, a more satisfactory explanation of these latter differences would appear to be a departure from the assumed conditions imposed by the equilibrium equations.

## VII. Conclusions

17. *Summary of Results.*—The line-reversal method of temperature measurement, as previously applied to the measurement of the temperature of stationary flames, has been adapted to measure the flame temperature in an internal combustion engine. Using an ordinary gasoline and moderate compression pressures, temperatures have been measured over a sufficient portion of the operating cycle to determine the maximum flame temperature during combustion, for varying air–fuel ratios within the combustion limits. These maximum flame temperatures have been compared with those computed for representative air–fuel ratios, considering the effect of residual exhaust

*Frank. Inst. Journ., Vol. 215, p. 149. (1933).
†Loc. cit., p. 34.

gas, variable specific heat and dissociation, but neglecting heat loss. The effect of spark timing on the flame temperature has also been determined for a limited range of operating conditions.

18. *Conclusions.*—The results of the investigation indicate that the following conclusions are justified:

(1) The flame temperature found from a line-reversal measurement is characteristic of thermal equilibrium, established throughout the gases in the engine cylinder early in the process of burning and maintained during the subsequent expansion.

(2) The maximum flame temperatures for a compression ratio of 3.86 to 1 have been found to vary from 3750 deg. F. abs., with the richest and leanest air–fuel mixtures, to 4450 deg. F. abs. with an air–fuel ratio of 13.9 to 1.

(3) The maximum observed flame temperatures and the corresponding calculated temperatures were found to be in closest agreement for the normal operating range of air–fuel ratios between 12 to 1 and 14 to 1. Throughout this range the calculated values are approximately 600 degrees higher than the observed values. With either richer or leaner mixtures the difference increases and reaches a maximum of 1000 degrees at the lean combustion limit.

(4) Since some of the discrepancy between the calculated and the observed maximum temperatures is undoubtedly due to experimental errors, the most likely of these have been considered and their upper limits estimated. If the observed values were increased by these amounts they would still be from 200 to 600 deg. F. below the calculated temperatures. It would, therefore, seem probable that this remaining difference is due to an inadequate analysis of the actual combustion process.

(5) The concentrations of the gases in an engine cylinder may be calculated when the temperature and pressure are known, hence the measurement of temperature, independent of the other thermodynamic variables, makes it possible to study the progress of the chemical reactions and equilibrium at high temperatures.

# VIII. APPENDIX

## Definitions and Nomenclature

The following definitions are for the terms and symbols used throughout this bulletin.

(1) $T$, True Temperature, is the temperature of a body as indicated by a suitable thermometer in contact and in equilibrium with it.

(2) $T_r$, Radiation Temperature of an imperfect radiator is the true temperature of a perfect radiator which is emitting the same total radiant energy, hence

$$T_r = \sqrt[4]{\epsilon} \cdot T$$

For a perfect radiator $\epsilon = 1$ and $T_r = T$

(3) $S_\lambda$, Brightness Temperature of an imperfect radiator is the true temperature of a perfect radiator which is emitting the same radiant energy in the spectral range $\lambda$ to $\lambda + d\lambda$. In specifying a brightness temperature the wave-length $\lambda$ must also be specified.

(4) $E$, Total Emissive Power, is the total radiant energy emitted in unit time by unit area of radiating surface into a solid angle $2\pi$.

(5) $E_\lambda$, Monochromatic Emissive Power, is the emissive power in the spectral range $\lambda$ to $\lambda + d\lambda$.

$$E = \int_0^\infty E_\lambda \, d\lambda$$

(6) $J$, Total Intensity, is the total radiant energy emitted in unit time by unit area of radiating surface into unit solid angle normal to the surface, then

$$E = \pi J$$

(7) $J_\lambda$, Monochromatic Intensity, is the intensity in the spectral range $\lambda$ to $\lambda + d\lambda$.

$$E_\lambda = \pi J_\lambda$$

(8) $\epsilon$, Total Emissivity, is the ratio of the total emissive power of an imperfect radiator to that of a perfect radiator at the same true temperature.

(9) $\epsilon_\lambda$, Monochromatic Emissivity, is the emissivity in the spectral range $\lambda$ to $\lambda + d\lambda$. The total and monochromatic emissivities may also be defined as the ratios of the corresponding intensities.

(10) $a$, Total Absorptivity, is the ratio of the total absorbed radiant energy to the total incident radiant energy. From Kirchhoff's Law

$$\frac{E}{a} = \mathbf{E}$$

when $E$ is the total emissive power of an imperfect radiation, $a$ the total absorptivity for the same, and $\mathbf{E}$ the total emissive power for a perfect radiator at the same temperature as the imperfect radiator.

(11) $a_\lambda$, Monochromatic Absorptivity, is the absorptivity in the spectral range $\lambda$ to $\lambda + d\lambda$.

(12) $r$, Total Reflectivity, is the ratio of the total reflected radiant energy to the total incident radiant energy.

(13) $r_\lambda$, Monochromatic Reflectivity, is the reflectivity in the spectral range $\lambda$ to $\lambda + d\lambda$.

The following terms occur in the discussion of the combustion reactions.

(14) $n$, Number of mols or pound-molecular weights of a single element or compound. The subscripts indicate the substance and state. Thus (1) indicates mols of $CO_2$ with complete combustion, (2) mols of $H_2O$ with complete combustion, $(e')$ mols of $O_2$ at equilibrium.

(15) $N$, Total number of mols in a mixture of elements and compounds. The subscripts indicate the state. Thus $(i)$ indicates the beginning of compression and

$$N_i = N_1 + N_c$$

where $N_1$ is the number of mols of entering charge of fuel and air, and $N_c$ is the number of mols of residual exhaust gas in the clearance space.

(16) $x$, Percentage of CO burned at equilibrium.

(17) $y$, Percentage of $H_2$ burned at equilibrium.

(18) $C$, Molecular Concentration of a substance in a mixture. In general

$$C_1 = \frac{n_1}{N}.$$

For a gaseous mixture

$$C_1 = \frac{p_1}{P},$$

where $p_1$ is the partial pressure of component (1) and $P$ is the total pressure.

(19) $K_p$, Equilibrium Constant. If the chemical reaction equation is written

$$\nu_1 A_1 + \nu_2 A_2 + \nu_3 A_3 + \ldots\ldots = 0$$

where $A_1$, $A_2$, etc., are the molecular formulae of the gases and $\nu_1$, $\nu_2$, etc., are the number of molecules of each taking part in the reaction, $\nu$ being positive when the substance is formed and negative when it is consumed, then

$$K_p = C_1^{\nu_1} \cdot C_2^{\nu_2} \cdot C_3^{\nu_3} \ldots\ldots$$

from the Law of Mass Action. The subscripts indicate the reaction involved. Thus (CO) indicates the reaction

$$-CO - \tfrac{1}{2}O_2 + CO_2 = 0,$$

($H_2$) indicates

$$-H_2 - \tfrac{1}{2}O_2 + H_2O = 0$$

and ($w.g.$) indicates

$$-H_2 - CO_2 + H_2O + CO = 0$$

the water-gas reaction.

(20) $H$, Heat of Combustion, is the lower heat of combustion at constant volume. The subscripts indicate the fuel. Thus (CO) indicates carbon monoxide, ($H_2$) indicates hydrogen and ($m$) indicates a fuel mixture.

(21) $U$, Intrinsic Energy of a mixture. The subscripts indicate the state. Thus (1) indicates the beginning of compression, (2) indicates the end of compression and (3) indicates the end of constant volume combustion.